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(54) **TORSION VIBRATION DAMPER**

(56)

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F16D 3/66

(52) **U.S. Cl.** **192/213.22**; 192/204

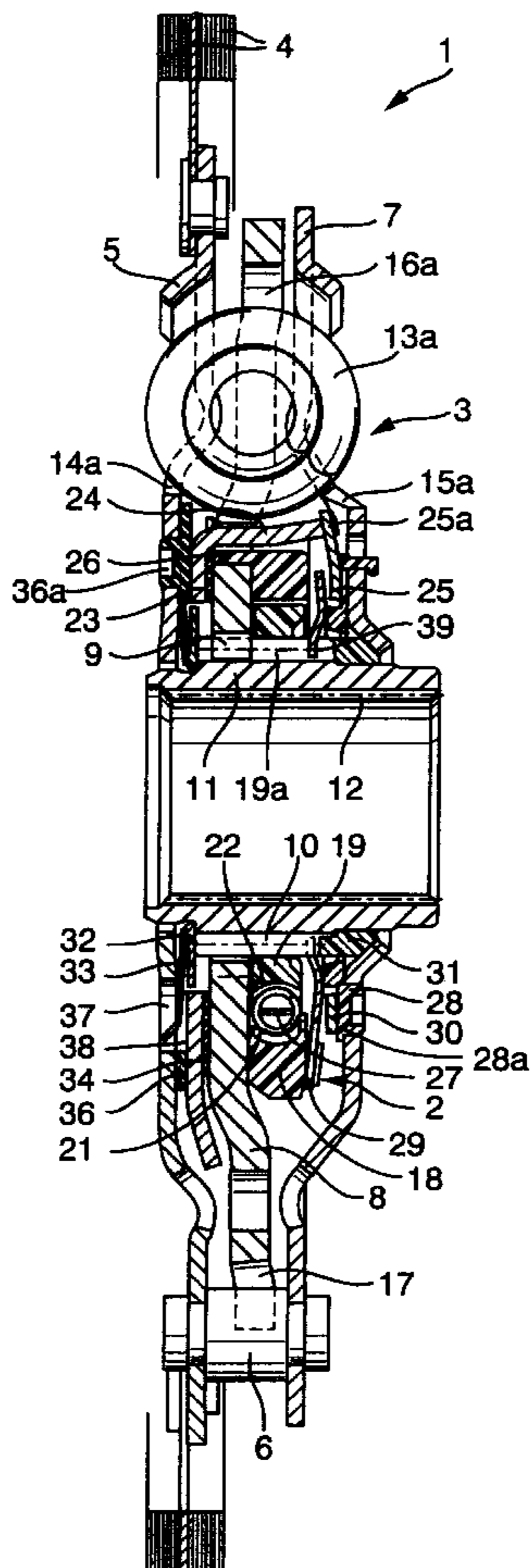
(58) **Field of Search** 192/213.22, 213.12,
192/204; 464/63, 68

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(57) **ABSTRACT**

The invention relates to a torsion vibration damper having a
main damper containing energy accumulators and a friction
device, as well as a preliminary damper containing energy
accumulators and a friction device.

54 Claims, 7 Drawing Sheets



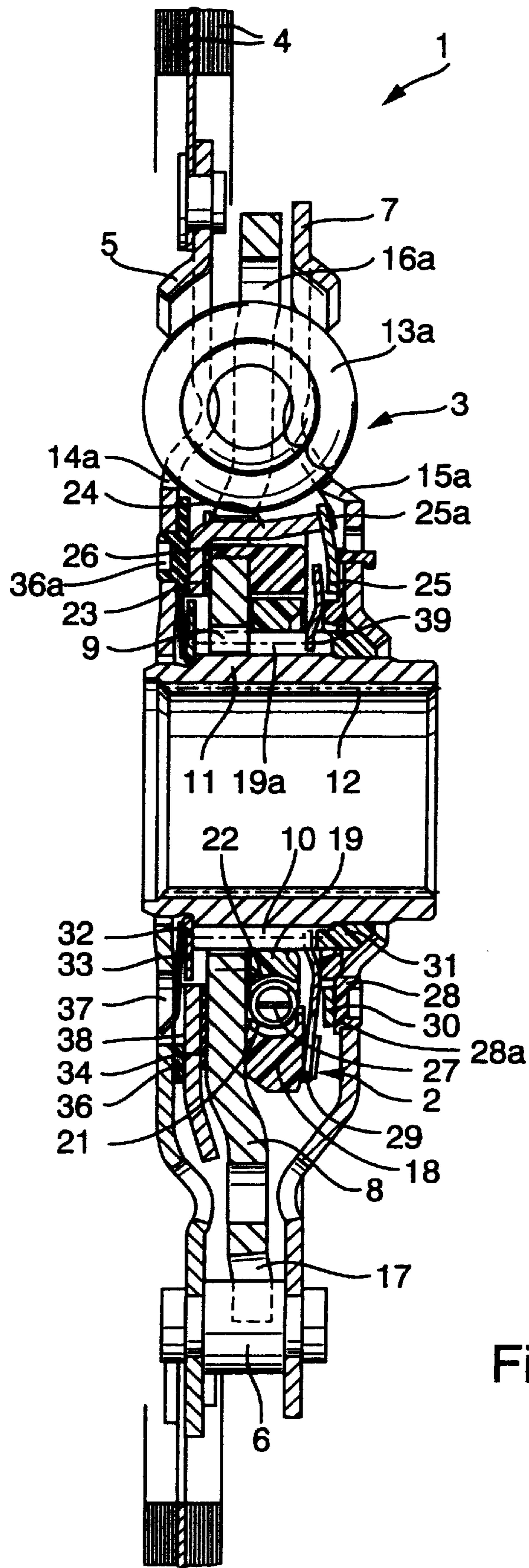


Fig. 1

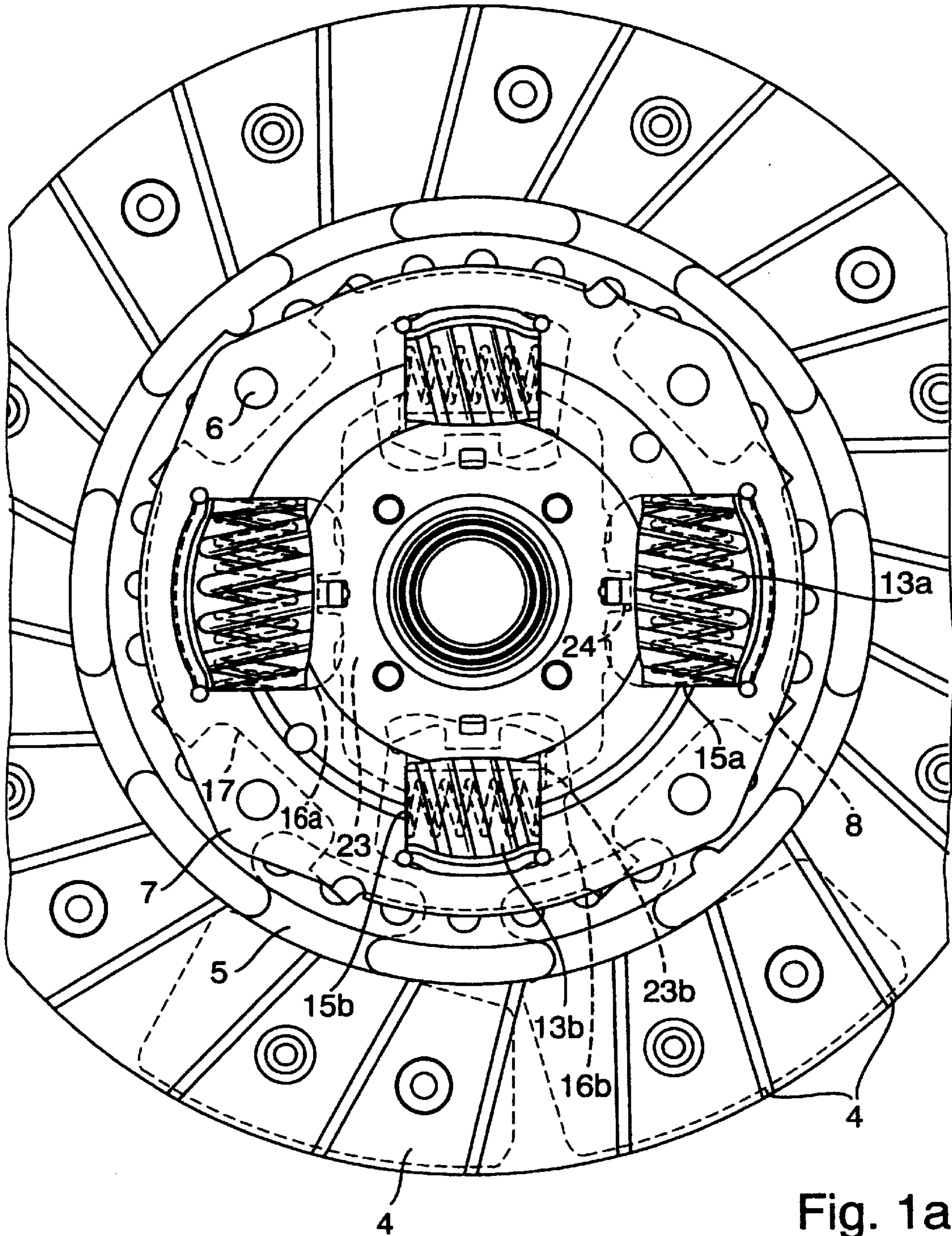


Fig. 1a

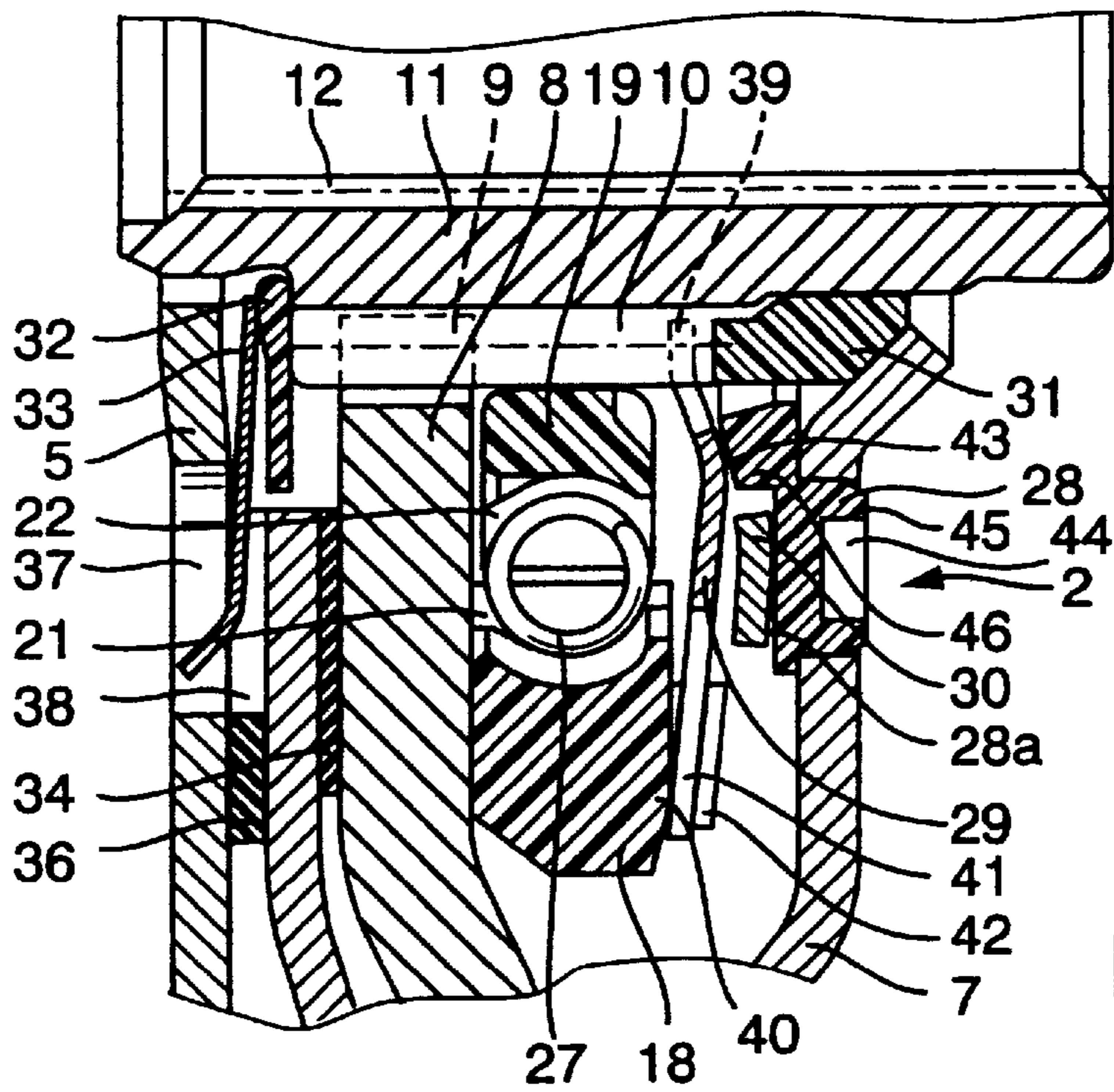


Fig. 2

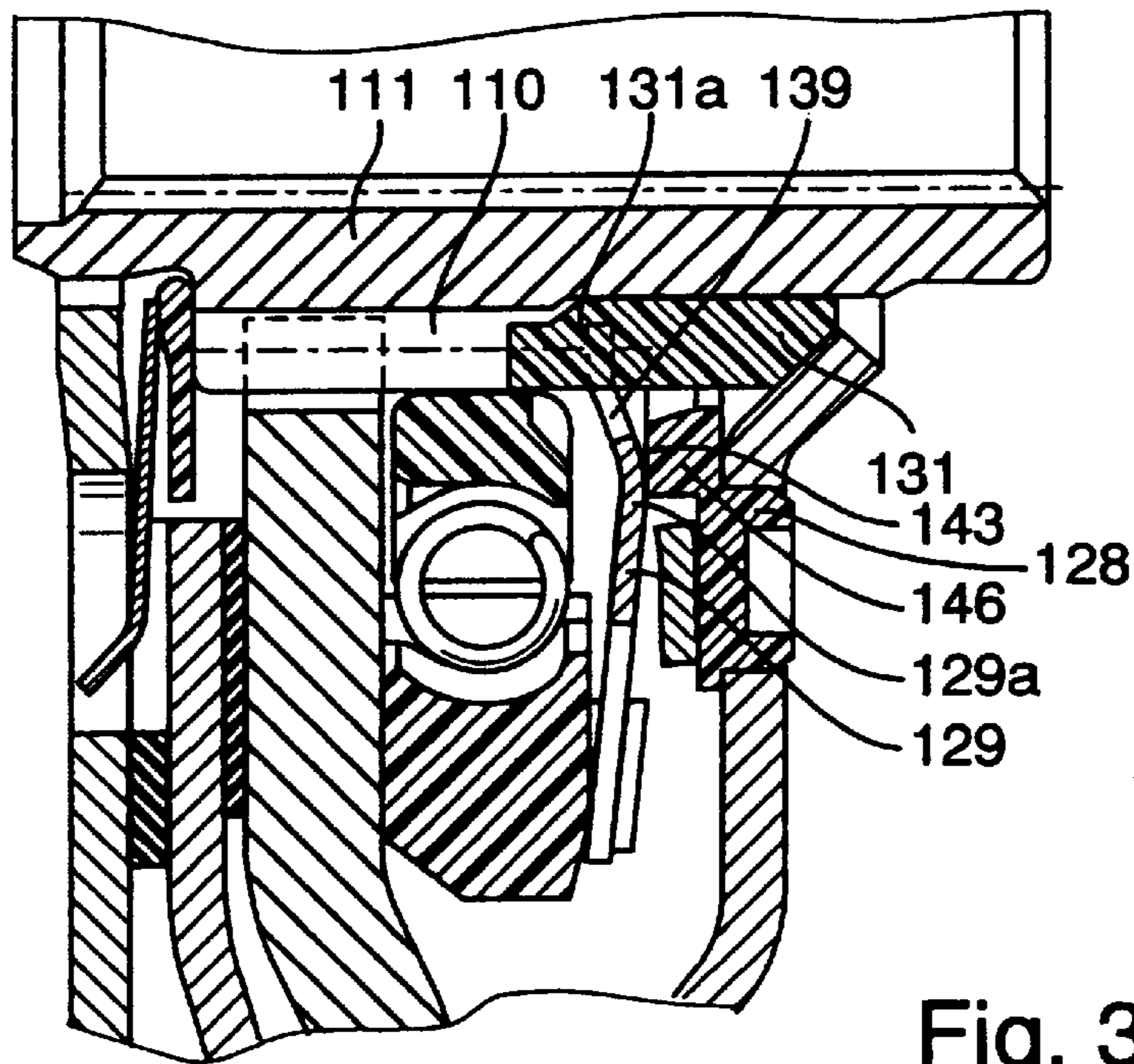


Fig. 3

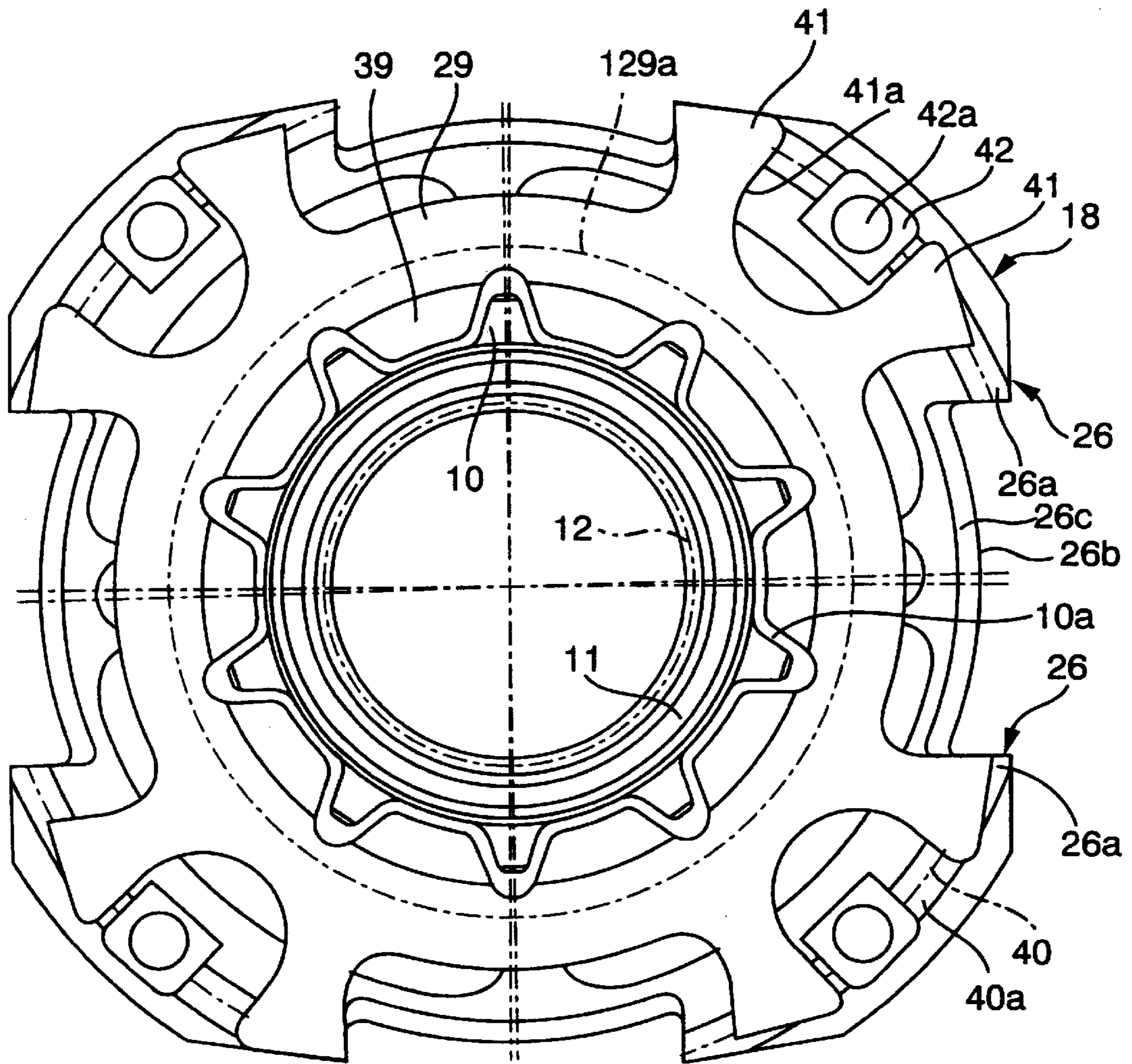


Fig. 4

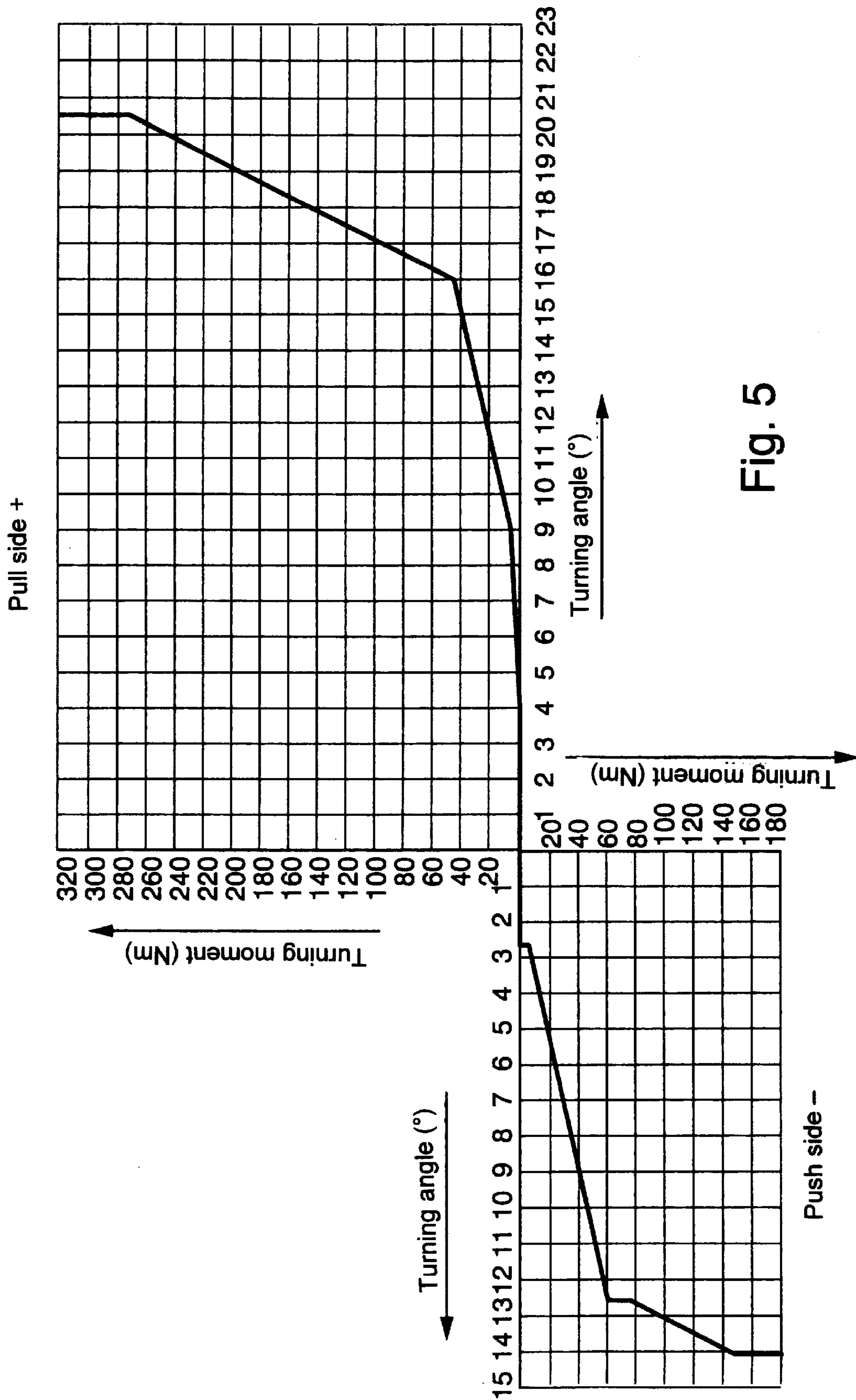


Fig. 5

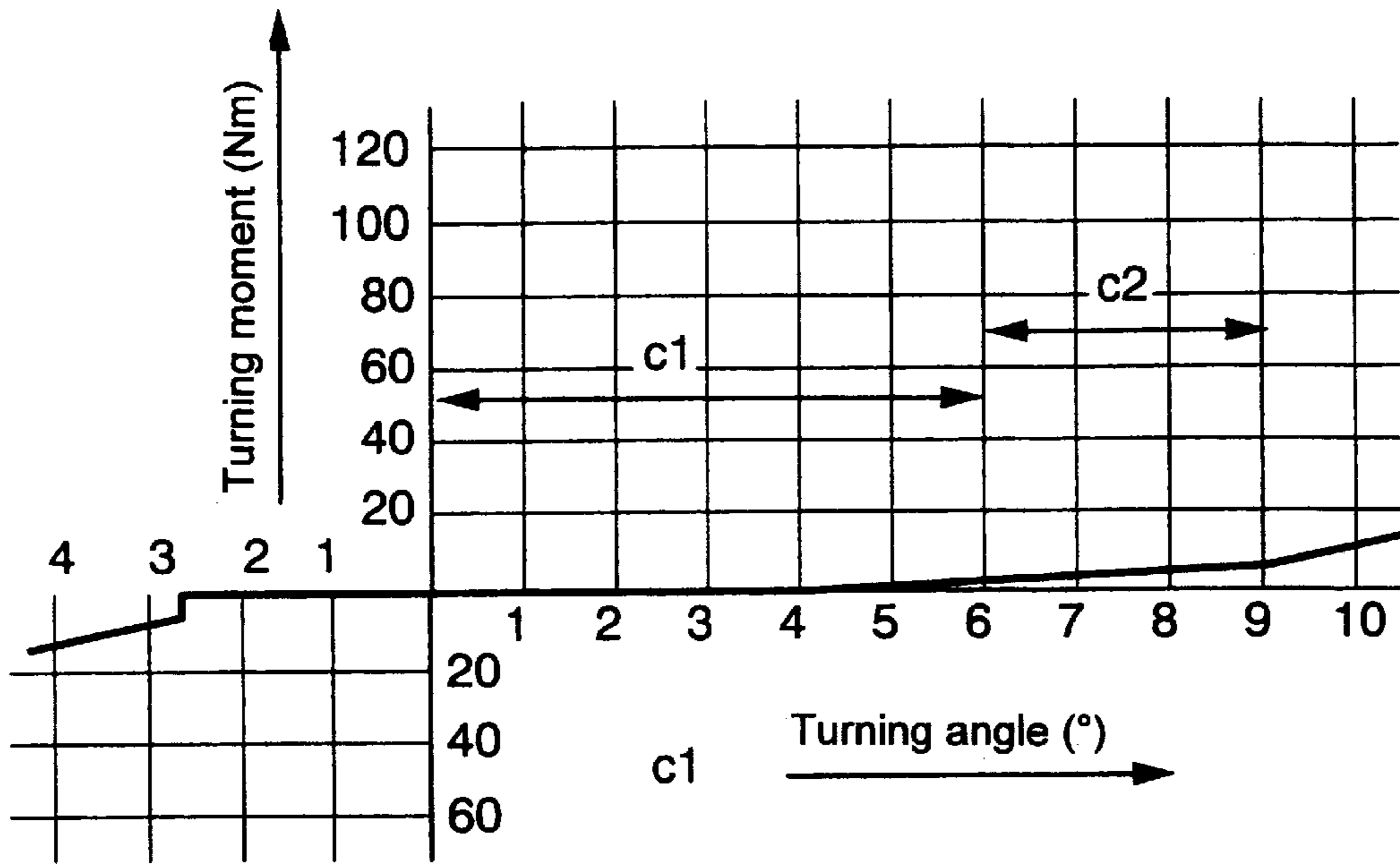


Fig. 6a

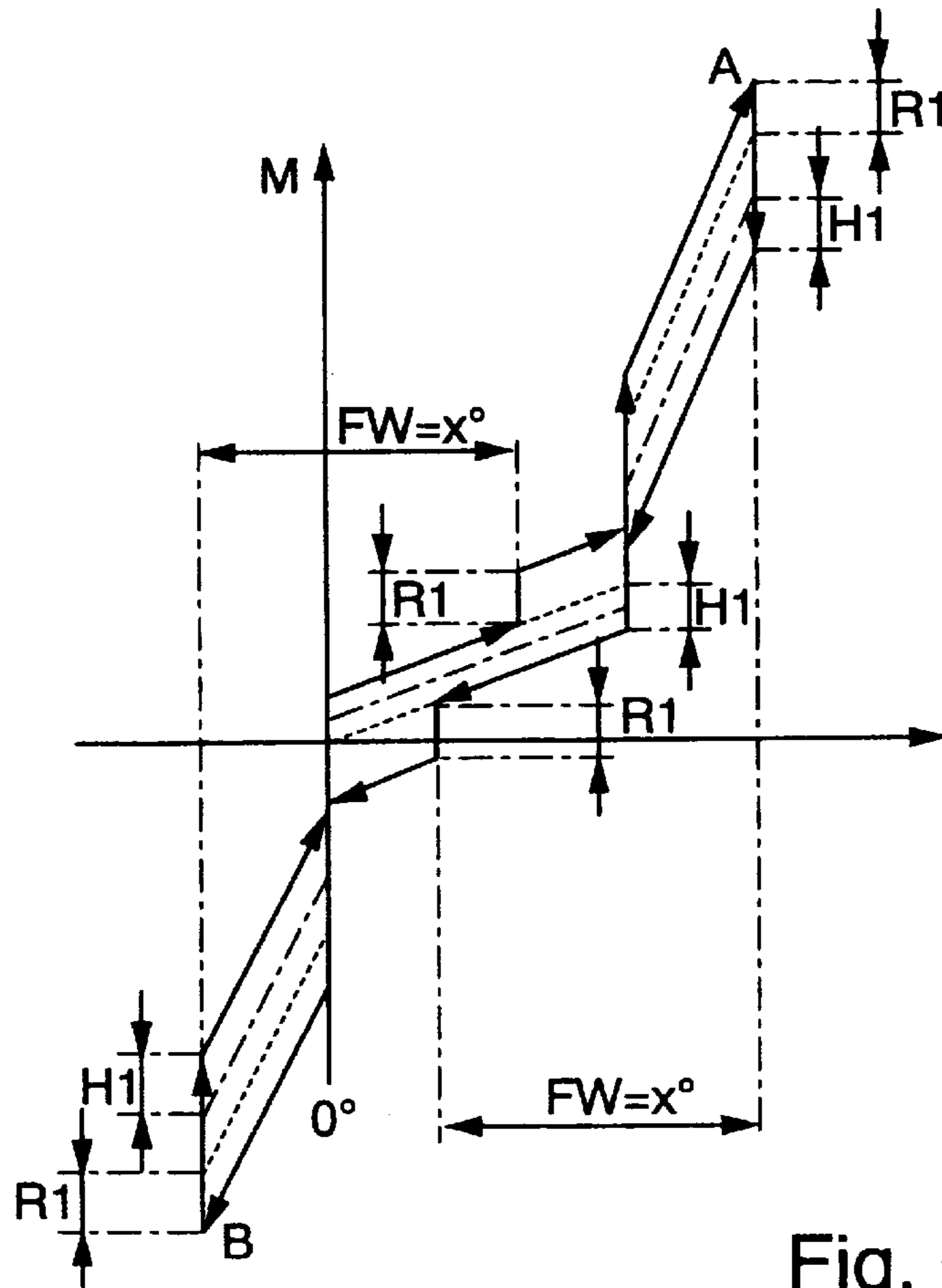


Fig. 6b

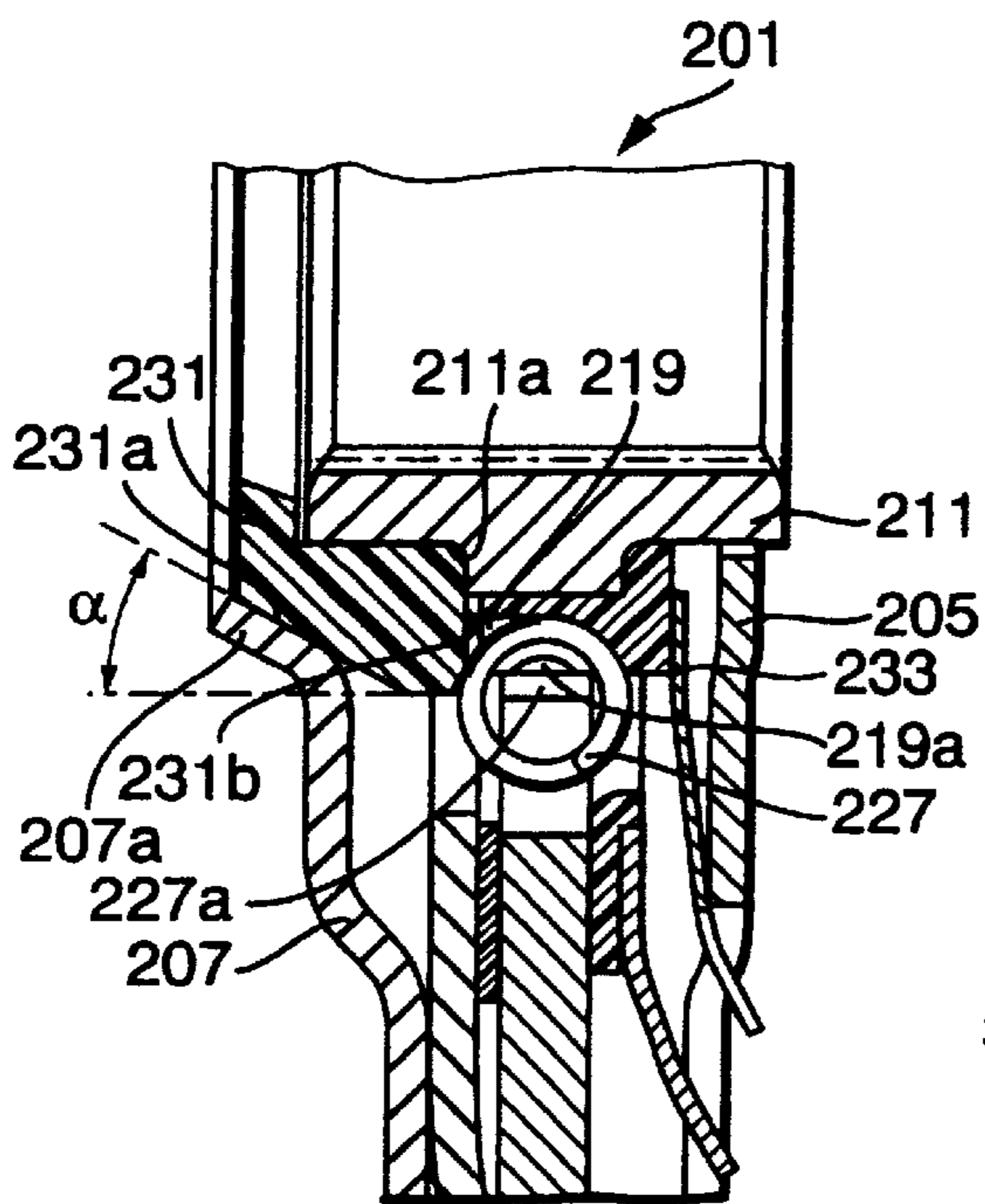


Fig. 7

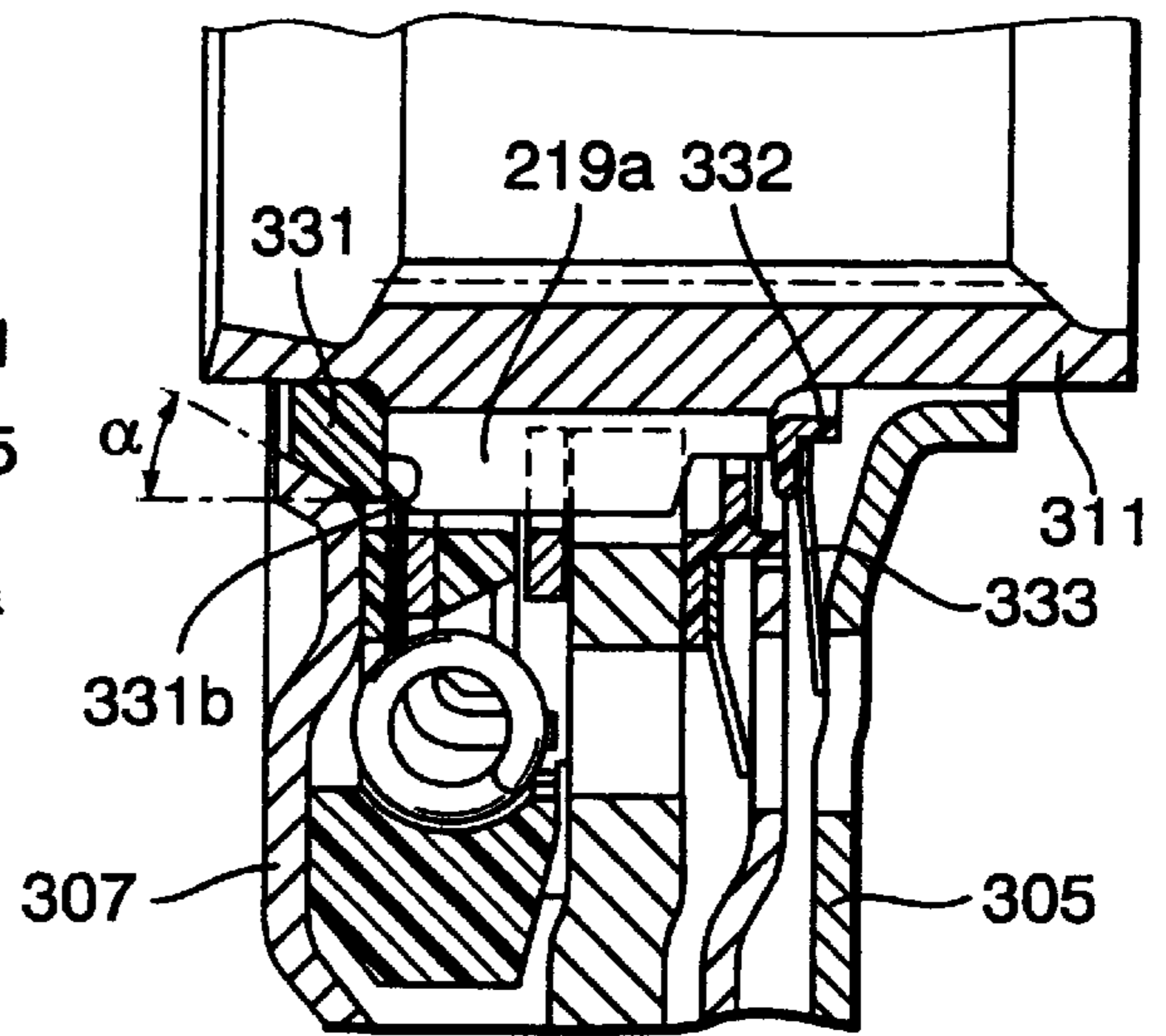


Fig. 8

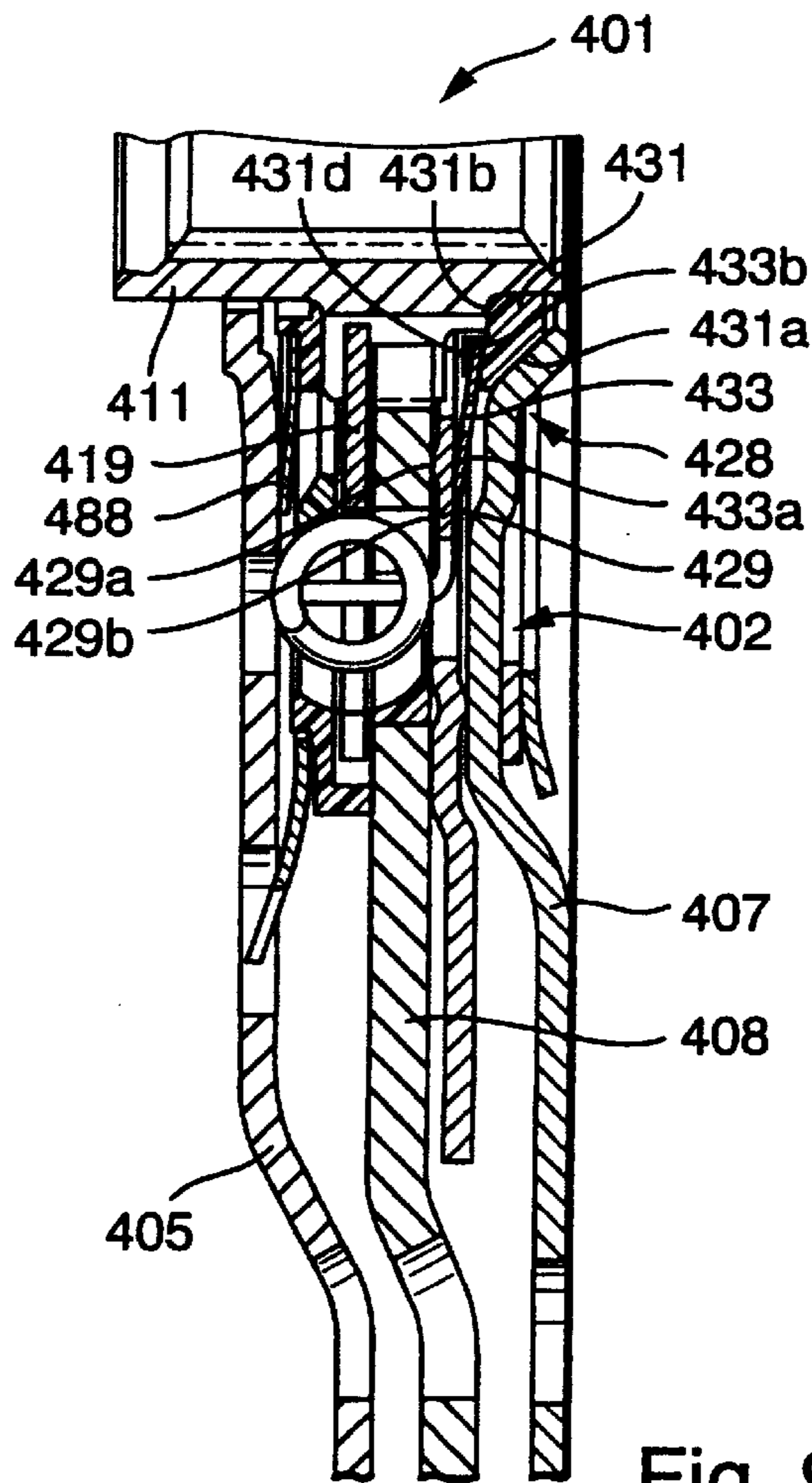


Fig. 9

TORSION VIBRATION DAMPER

This is a continuation of International Application No. PCT/DE99/01346, filed May 4, 1999, the entire disclosure of which is hereby incorporated by reference.

The invention relates to a torsion vibration damper, more particularly for motor vehicle clutch discs, with at least one preliminary damper acting in a predetermined angular area and having energy accumulators of lower stiffness, and at least one main damper acting in a predetermined angular area and having energy accumulators of greater stiffness, wherein the energy accumulators are active between the respective input and output parts of the preliminary and main dampers, and the output part of the torsion vibration damper is a hub provided with inner profiled sections to fit onto a gearbox shaft, as well as a flange forming the output part of the main damper is provided with inner profiled sections so that the inner profiled sections engage with the outer profiled sections of the hub and through this profiling the flange of the main damper is able to execute restricted relative rotation relative to the hub, as well as having at least one disc part which forms the input part of the main damper and holds the friction linings, and with at least one friction device.

Torsion vibration dampers with preliminary and main dampers having associated friction devices are known for example from DE 40 26 765 which each have a separate friction device for the main and for the preliminary damper, wherein the preliminary damper has a two-stage friction build-up and two-stage mounted energy accumulators for adapting to the different conditions. The drawback with this type of torsion vibration damper is the inability to dampen by simple means the torsion vibrations of the pressure plate at high speeds, such as occur for example during engagement and disengagement processes, so that the turning path of the preliminary damper is exceeded and the preliminary damper strikes against its restricting stop and thereby causes clutch noises which cannot be tolerated. Furthermore a construction of this kind is relatively complicated and the assembly becomes correspondingly expensive through the numerous structural elements used, which is all the more apparent if additional measures have to be used to counteract the clutch knocking previously described.

The object of the present invention is therefore to produce a torsion vibration damper of the kind mentioned above which provides the possibility of damping large torsion vibration amplitudes at high speeds, uses a minimum number of component parts and allows simple assembly.

According to the main claim this is achieved in that a torsion vibration damper, more particularly for motor vehicle clutch discs, is provided having at least one preliminary damper acting in a predetermined angular area and having energy accumulators of lower stiffness, and at least one main damper acting in a predetermined angular area and having energy accumulators of greater stiffness, wherein the energy accumulators are active between the respective input and output parts of the preliminary and main dampers, and the output part of the torsion vibration damper is a hub provided with inner profiled sections to fit onto a gearbox shaft, as well as a flange forming the output part of the main damper is provided with inner profiled sections so that the inner profiled sections engage with the outer profiled sections of the hub and through this profiling the flange of the main damper is able to execute restricted relative rotation relative to the hub, as well as having at least one disc part which forms the input part of the main damper and holds the friction linings, at least one friction device, wherein a spring

which engages in the outer profiled sections of the hub is provided and controls at least one part of the friction device and defines the friction engagement.

It is thereby advantageous to make the hub two-part wherein an additional hub part with outer profiled section can hold the inner profiled section of the spring, as well as to provide a restricted relative rotation forming a free angle between the spring and the hub whereby the spring is entrained with the input part and thus a friction moment does not occur in the normal active area of the preliminary damper, that is friction is delayed until the free angle is used up and a high friction gradient, a so-called friction jump, ensues through the inner profiled section of the spring stopping against the outer profiled section of the hub.

Furthermore it is advantageous to set up the relative rotation between the spring and hub so that a delay with a free angle α is caused between the spring and hub, wherein this free angle α is the range from $\pm 2^\circ$ to $\pm 3^\circ$, preferably $\pm 2.5^\circ$.

To perform its function as a control element of the friction device the spring has in a preferred embodiment an inner profiled section complementary with the external profiled section of the hub disc and forming teething with the external profiled section of the hub disc to thereby allow the said free angle.

Advantageously one design form has the preliminary damper arranged with its input and output part so that the output part of the preliminary damper is connected rotationally secured with the hub and the spring is tensioned between the input part of the preliminary damper and the disc part and/or a component part fixedly connected thereto. For design reasons in a further embodiment it is proposed that the previously mentioned component part connected fixed to the disc part is a second disc part set apart by spacer bolts and on which to optimise the friction coefficients a friction ring is fixed with which the spring forms the friction face.

The advantageous construction of a further embodiment of the spring is produced where the spring has an external profiled section with at least one radially outwardly pointing tongue wherein advantageously several tongues are provided which are spread out over the circumference and which have radially outwards an approximately semicircular recess. Double the number of friction tongues are now produced which form on the preliminary damper which is preferably designed as a friction face, an additional friction surface between the spring and preliminary damper.

In a further advantageous design the tongues are widened out at their radial outer side so that the friction face between the spring and tongues can be enlarged and the friction can thereby be improved.

Further advantageous design possibilities for optimising the friction surface between the spring and input part of the preliminary damper are offered wherein the input part of the preliminary damper has, on the axial side facing the spring in the area of the contact surface between the input part and the spring tensioned with a contact bearing angle β , a rounded end which has a pitch angle such that the bearing angle β of the spring is approximately $\beta=0$.

A further advantageous design relates to the input part of the preliminary damper which has at least one axially extending pin on the axial side facing the spring, wherein an arrangement of several pins spread uniformly over a circumference of constant diameter is advantageous and whose number corresponds to the recesses of the tongues formed on the outer circumference of the spring. It is further advantageous if the pins engage with play in the recesses of

the tongues and thus serve for pre-centring during assembly. The play between the tongues and pins is thereby advantageously greater than the free angle of the teething between the spring and hub, so that the control of the friction device is not impeded. The pins can in further embodiments serve as stops for restricting the spring path.

Further advantageous design possibilities relate to the friction ring connected to the disc part and which in one design model is advantageously formed so that it is press-fitted into a bore provided in the disc part through at least one, and preferably several axially extending hollow pins spread evenly over the circumference, so that the friction ring is fixed on the disc part during assembly and is connected rotationally secured to the disc part.

In a further embodiment the friction ring has a ring located on the outer circumference and raised axially in the direction of the spring whereby its ring face drops advantageously towards its inner diameter and the ensuing ring face forms relative to the inner face of the ensuing ring a phase angle γ which is advantageously designed so that the contact bearing angle β of the spring on the friction ring is approximately $\beta=0$ and thus an improved friction surface is formed. The formation of the raised ring has the advantage that radially outside the outer circumference of the raised ring of the friction ring there is a further plate spring which belongs to a friction device of the main damper and which thereby is formed without taking up additional axial structural space. It is supported on one side on the non-raised inner ring face of the friction ring and on the other side on the axially aligned tabs of the control plate for the second stage of the main damper so that the friction ring forms at least a part of the friction device of the preliminary and main damper.

A further development of the invention relates to the arrangement and design of the preliminary damper for space-saving fitting of the spring engaging in the hub. Advantageous for this is an arrangement where the preliminary damper is housed axially between the disc part and a second associated disc part so that the spring can be tensioned directly between one of the two disc parts or a friction ring attached thereon and the preliminary damper. In principle however other design forms are possible wherein the preliminary damper has an axial stagger relative to the main damper and the spring is tensioned between the first disc part or a component part connected thereto and the input part of the preliminary damper. Furthermore the first disc part can be attached in the axial direction centrally on the hub whereby the preliminary damper and flange can be mounted axially on the same side or the disc part can be mounted flanked on both sides.

To fix the output part of the preliminary damper on the output part of the main damper in one embodiment it is proposed that pins attached to the output part of the preliminary damper are fitted into windows provided in the output part of the main damper for receiving the energy accumulators. These pins are provided complementary with the two radially inner corners of each window on the input part of the preliminary damper, are shaped out axially and are engaged in the corners of the windows. They simultaneously centre the preliminary damper on the output part of the main damper.

A further embodiment according to the invention relates to the design of the hub wherein the external profiled section of the hub is continued in a cone which has for this purpose a positive-locking inner profiled section or an axially mounted positive-locking profiled section and wherein the spring engages by its inner profiled section in an external

profiled section of the cone. This solution produces a significant advantage with regard to assembly since through the variation of the cone which is simple to produce different free angles of the spring can be created without changing the hub or spring.

The invention will now be explained in further detail with reference to the embodiments shown in the accompanying FIGS. 1 to 9 in which:

FIG. 1 shows the torsion vibration damper in longitudinal section;

FIG. 1a shows a partial view of the torsion vibration damper;

FIG. 2 shows a longitudinal sectional view of a part of FIG. 1 relating to the preliminary damper;

FIG. 3 shows a longitudinal sectional view of a part of another embodiment relating to the preliminary damper;

FIG. 4 is a view of the input part of the preliminary damper with fitted spring;

FIG. 5 shows a characteristic line of an embodiment;

FIG. 6a shows a characteristic line of the preliminary damper whilst omitting the friction jump;

FIG. 6b shows the path of the friction moment for rotation over the entire active area of the preliminary damper with friction jump; and

FIGS. 7 to 9 show in detail further embodiments of torsion vibration dampers.

The torsion vibration damper 1 shown in the drawings has a preliminary damper 2 and a main damper 3. The input part of the torsion vibration damper 1 which represents the input part of the main damper 3 is formed by a first disc part 5 (not shown complete) supporting friction linings 4 as well as by a second disc part 7 connected rotationally secured to the first part by spacer bolts 6. The output part of the main damper 3 is formed by a flange 8 which has an internal profiled section, preferably internal teeth 9, which engage in an external profiled section, preferably external teeth 10, of a hub 11. Between the external teeth 10 of the hub 11 and the internal teeth 9 of the flange there is a tooth flank play in the circumferential direction which corresponds to the active area of the preliminary damper 2. The hub 11 furthermore has internal teeth 12 for fitting rotationally secured but axially displaceable on a gearbox input shaft.

The main damper 3 has a first set of coil compression springs 13a which can comprise a pair of coil compression springs boxed in each other, for the first main damper stage, which are provided in window-shaped recesses 14a, 15a of the first and second disc part 5, 7 on one side as well as in window-shaped cut-out sections 16a of the flange 8. The action of the coil compression springs 13a is activated through the relative rotation of the recesses 14a, 15a opposite the recesses 16a, after using up the free angle, in which the preliminary damper is active, between the hub 11 and flange 8. A second set of coil compression springs 13b (FIG. 1a) of higher stiffness, which can also consist of coil compression springs boxed in each other, but off-set on a circumference of the same diameter by an angle of preferably 90° relative to the coil springs of the first stage, for the second main damper stage is set in the recesses 14b, 15b (FIG. 1a) of the disc parts 5, 7 and in the window-shaped recesses 16b (FIG. 1a) of the flange 8 wherein the recesses 16b have a larger cut-out section than the length of the coil compression springs 13b whereby during relative rotation of the disc parts 5, 7 opposite the flange 8 the action of this coil spring set 13b is only used with greater turning angles and thus a second damper stage of the main damper is formed. Between the flange 8 and the disc part 5 is a friction control part 23 which has recesses 23a (FIG. 1a) for housing the coil

spring set **13b** (FIG. 1a) and on these recesses **23a** axially aligned tabs **23b** (FIG. 1a) which engage in the flange **8** and during rotation of the flange **8** about a turning angle which activates the second main damper stage, entrain the friction control part **23** whereby a friction engagement which only acts in the second main damper stage is produced on a friction disc **34** attached between the friction control part **23** and the flange **8**. Furthermore the friction control part **23** has axially extending tabs **24** for holding a plate spring **25** which is supported on a further friction ring **28** fixed on the disc part **7** and thus defines the friction engagement on the friction discs **28** and **26**. The rotation of the main damper is restricted by the spacer bolts **6**, which connect the two disc parts **5** and **7** together, stopping against the end contours of the cut-out sections **17** of the flange **8** into which they project axially.

The preliminary damper **2** is mounted axially between the flange **8** and the disc part **7**. The input part **18** made from plastics preferably by injection moulding is connected rotationally secured to the flange **8** through pins **26** projecting axially into the corners of the recesses **16** of the flange **8**. The output part **19** of the preliminary damper **2** made from plastics preferably by means of injection moulding is connected rotationally secured through internal teeth **19a** to the external teeth **10** of the hub **11** whereby as a result of the tooth flank play of the internal teeth **9** of the flange **8** and the external teeth **10** of the hub **11** relative rotation is possible between the output part **19** and input part **18** level with the active area of the preliminary damper **2** against the action of the coil compression springs **27** housed in the window-shaped recesses **21**, **22** in the output part **19** and the input part **18**. The recesses **22** of the output part **19** provided for controlling the coil compression springs **27** are divided alternately into two groups on a circumference of constant diameter of the preliminary damper **2** whereby the recesses of the one group arranged on the same circumference are formed longer in the circumferential direction compared to the other group whereby the coil compression springs **27** housed in this group are only controlled in the event of greater relative rotations to thereby form a second preliminary damper stage. It is advantageous if the coil compression springs **27** belonging to this group have at the same time a higher stiffness.

The friction device of the torsion vibration damper **1** is made up as follows: the basic friction of the main damper **3** takes place through friction engagement of the friction control disc **23** and the disc part **5** on the friction disc **36** which is connected rotationally secured to same by means of hollow pins **36a**, wherein the friction engagement takes place over the entire active area of the main damper **3** and the spring **29** which is supported on the friction ring **28** and on the input part **18** of the preliminary damper **2** which is supported in turn on the flange **8** defines the friction moment. The friction moment of the friction disc **34** already mentioned above and acting in the second main damper stage, between the friction control part **23** and the disc part **5**, is likewise fixed by the plate spring **30** which is supported on the friction control part **23**. This is joined by a friction moment arising at the friction disc **28** and acting in the entire active area of the main damper **3** and which is defined by the plate spring **29** which is supported on the input part **18**, formed as a friction ring, of the preliminary damper **19**. After using up the free angle which the spring **29** forms during engagement of its internal teeth **39** with the external teeth **10** of the hub **11**, the friction also becomes active in the preliminary damper **2** which leads to a delayed friction jump in the preliminary damper **2**. The basic friction of the

preliminary damper arises at the friction disc **32** which adjoins the inner circumference of the friction disc **36** and is pressed against the hub **11** by means of a plate spring **33** supported on the disc part **5** and provided with tooth-like outer profile, whereby a part of the radially longer formed teeth engages on one side in recesses **37** of the disc part **5** and thereby produces the rotational lock of the spring and on the other side the remaining part of the shorter teeth engages in recesses **38** of the friction disc **36**, wherein the hub **11** is supported in turn on the disc part **7** by means of a cone **31**.

The cone **31** which is provided with axial recesses **31a** for keyed engagement with the external teeth **10** of the hub **11** serves to centre the disc part **7** on the disc part **5** and causes the fixing of the friction force on the friction discs **34** and **36**.

FIG. 1a shows the torsion vibration damper **1** according to the invention in partial view from which for clarity the preliminary damper has been omitted and the parts arranged underneath the disc part **7** are shown by dotted lines. The parts described above are in detail: the first disc part **5** with the friction linings **4** which have grooves **4a** is connected rotationally secured by the retaining bolts **6** to the second disc part **7**, in-between—building up from below—are the friction control part **23** with its two groups of tabs **23b** and **24** as well as the recesses **23a** for the second spring set with the coil compression springs **13b** which are also fitted into the recesses **14b**, **15b** of the two disc parts **5**, **7**. The first spring set with the coil compression springs **13a** is housed in the recesses **14a**, **15a** of the two disc parts **5**, **7**. The flange **8** undertakes the control of the spring sets **13a**, **13b** by its recesses **16a**, **16b** for the two coil compression spring sets **13a**, **13b** in the turning angle of the main damper **3** defined by the recesses **17** and retaining bolts **6**, wherein the recesses **16b** have a larger cut-out section than the length of the coil compression springs **13b** so that the entrainment of the springs **13b** thereby forming a second main damper stage only takes place in the event of a larger turning angle.

A more detailed explanation of the preliminary damper **2** with the component parts enclosing same is shown in the section from FIG. 1 provided in FIG. 2. The spring **29** according to the invention is tensioned between the friction ring **28** and the input part **18** of the preliminary damper **2**. The inner circumference of the spring **29** is designed as an inner profiled section, preferably as internal teeth **39**, which engages in the external profiled section, preferably external teeth **10** of the hub **11** and has a circumferentially arranged tooth flank play which allows relative rotation between the hub **11** and spring **29**. The tooth flank play is selected so that the turning angle is smaller than the active area of the preliminary damper **2** so that in the case of large turning angles of the preliminary damper the friction arising through the friction faces **40a** (FIG. 4) between the spring **29** and input part **18** of the preliminary damper on one side and between the spring **29** and friction ring **28** on the other side, after using up the free angle set between the teeth **10**, **39**, becomes active on the preliminary damper and produces a friction jump wherein prior to using up the free angle the spring co-rotates on the input part **18** without producing friction moments.

On the outer circumference the spring **29** has evenly distributed tongues **41** with approximately semi-circular shaped recesses **41a** (FIG. 4) into which axially protruding pins **42** of the input part **18** project with a play which does not impede the rotation of the spring **29** in the free angle provided but allows assistance during assembly. The input part **18** is formed at the friction face **40a** (FIG. 4) with the spring **29** as a rounded end **40** so that the spring **29** adjoins

with the smallest possible contact bearing angle β and thus the friction face **40a** (FIG. 4) is optimized.

The friction ring **28** forms with the spring **29** a designated friction surface **43** of a raised ring **46** whereby the ring surface drops in the direction of the internal diameter of the ring in order to produce a small contact bearing angle β . On the inner circumference of the friction ring **28** which is fitted rotationally secured by axially formed hollow pins **45** in recesses **44** of the disc part **7**, a plate spring **30** having outwardly extending tabs **25a** (FIG. 1) attached on the circumference and supported by these tabs **25a** against the tabs **24** of the friction control part **23** (FIG. 1) adjoins the ring face **28a** and causes a friction moment acting on the main damper **3**.

A further design model is shown in the form of a longitudinal section in FIG. 3. A torsion vibration damper **101** according to the invention and similar to the torsion vibration damper **1** has a hub **111** with axially shorter external teeth **110** in which the cone **131** as a second hub part engages with positive locking by means of axial teeth. Furthermore the cone **131** supports external teeth **131a**, which preferably differ from the external teeth **110** of the hub **111**, into which the spring **129** engages by means of internal teeth **139** producing the tooth flank play required for the delayed friction, whereby it is not necessary to adapt the spring **129** to the hub **111** and in the case of different requirements regarding the delayed friction system only the cone **139** need be changed in respect of the free angle which is to be varied.

A further design possibility relates to the friction ring **128** whose raised ring **146** has a flat ring face **143** wherein the friction face between the ring **146** and spring **129** is optimized in that on the spring **129** in the area of the contact surface with the ring **146** the circumferential bend **129a** is adapted to the path of the friction face **143**.

FIG. 4 shows the hub **11** having the internal teeth **12** which engage in the external teeth of a gearbox input shaft (not shown), and the external teeth **10** which engage with tooth flank play **10a** in the internal teeth of the spring **29** whereby through a tooth flank play **10a** in the circumferential direction of preferably $\pm 2.5^\circ$ the friction jump is controlled by means of the friction moment arising on the friction faces **40a** between the spring **29** and the input part **18** of the preliminary damper **2** on one side and between the spring **29** and the friction ring **28**, **128** (FIGS. 1, 2 or 3) on the other side, wherein the size of the friction moment is fixed by the axially acting spring rate of the spring **29**.

The spring **29** has on its axial circumference radially extending tongues **41** which hold through their approximately semi-circular shaped recesses **41a** the pins **42** which are formed with an axially aligned centre bore **42a** wherein the play required for smooth setting the friction jump remains between the tongues **41** and pins **42**. The pins **42** serve as stops against the direction of rotation.

The tongues **41** are widened out at their outer side so that additional friction surface is obtained which is optimized through a rounded end formation **40** of the input part **18** of the preliminary damper **2** relative to the contact bearing angle β of the spring **29** with the input part **18**.

Fixing the preliminary damper **2**, which is shown here without the output part **10** and the coil springs **27** (FIG. 1), on the flange **8** is undertaken by means of pins **26** extending axially at the corners **26a** at the side remote from view and which fit into the window shaped recesses **16a**, **16b** of the flange **8** (FIG. 1). The edges **26c** of the recesses **26b** of the input part **18** of the preliminary damper **2** extending down in the axial direction thereby form a positive locking connection with the window shaped recesses **16a**, **16b** of the flange.

FIG. 5 shows the theoretical path of the turning moment in dependence on the turning angle. The path of the turning moment in the event of small turning angles in the direction of the pull side, thus in the direction in which the drive assembly turns the torsion vibration damper whilst the gearbox input shaft is still stationary is in this embodiment orientated up to about 9° from the damping properties of the two-stage preliminary damper **2** (FIG. 6a). The first stage of the main damper **3** is set after using up the free angle between the external teeth **10** of the hub **11** and the internal teeth **9** of the flange **8**. The second main damper stage is set after using up the clearances of the recesses **16b** of the flange with a turning angle of 16° . The increase in the turning moment is more than double the turning moment of the first main damper stage since the coil compression springs **13b** of the second main damper stage have a higher stiffness compared with the coil compression springs **13a** of the first stage. With a turning angle of about 20.5° in this embodiment the recess **17** of the flange **8** strikes against the retaining bolts **6** which connect the disc parts **5**, **7** together so that the action of the main damper stage is terminated.

In the push side direction the free angle of the preliminary damper **2** is restricted to a turning angle of 2.5° so that the first main damper stage only starts from this turning angle. Also the start of the action and the stop of the second main damper stage are restricted to turning angles of 12.5° and 14° respectively.

FIG. 6a shows an enlarged area of FIG. 5 for a better illustration of the turning moment of the preliminary damper **2** over the turning angle. In the pull direction (right section of the drawing) the first preliminary damper stage **c1** operates at turning angles up to 6° . In the case of larger turning angles the clearance of the recesses **22** of the output part **19** of the preliminary damper **2** is used up and the second preliminary damper stage **c2** is activated up to an angle of 9° at which the free angle between the external teeth **10** of the hub **11** and the internal teeth **9** of the flange **8** is used up and the main damper device is used. The method of operation of the preliminary damper is in this embodiment serial, that is the spring tension of the preliminary damper **2** remains during the action of the main damper **3**. The preliminary damper **2** has during the push operation a restricted turning ability, namely a turning angle of 2.5° wherein only the first preliminary damper stage is activated.

FIG. 6b shows the path of the turning moment **M** of an embodiment of the preliminary damper **2** according to the invention in dependence on the turning angle α taking into account the hysteresis **H1** conditioned by the friction device.

The solid lines marked by arrows thereby show the path of the turning moment in the direction of the arrows when the preliminary damper **2** has turned and with a reversal of the turning angle, the dashed lines show the path of the curve of the turning moment without friction jump and the chain-dotted line shows the mean value of the turning moment corrected by the hysteresis without taking into account the friction jump. Starting with a turning angle α at which during push operation the preliminary damper is standing against the stop and only the first preliminary damper stage is active, the turning moment **M** in relation to the pull side decreases to a turning angle of 0° , the neutral phase of the first preliminary damper stage. The turning moment **M** then increases successively in dependence on the spring rate and the basic friction of the first preliminary damper stage up to the free angle **FW** between the internal teeth **39** of the spring **29** and the external teeth **10** of the hub **11**. The spring **29** is then entrained by the hub **11** and produces through the resulting relative rotation a friction moment at the contact

faces with the friction ring **28** and with the input part **18** of the preliminary damper **2** from which the illustrated friction jump **R1** arises with the turning angle FW . The additional friction moment is superimposed on the friction moment of the first preliminary damper stage until the second preliminary damper stage by way of example **c2** (FIG. **6a**) is activated with an additional friction moment in the case of a turning angle of 6° . From the pitch of this curved section it is clear that the coil compression springs of the first preliminary damper stage have smaller stiffness than the coil compression springs of the second preliminary damper stage. At the end **A** of the active area of the preliminary damper **2** in the pull direction the turning angle is changed over whereby the hysteresis H_i acts in the reverse direction and the friction moment of the friction jump **R1** ceases since now the relative rotation of the spring **29** opposite the hub **11** is provided again through the changed direction of rotation by means of the free angle relative to the hub **11**. With a reverse turning angle of 3° in relation to the embodiment in FIG. **6a** the second preliminary damper stage becomes inactive again and the friction moment M drops to the value of the first damper stage reduced by the hysteresis H_1 in the event of a full deflection. A further decrease in the turning angle α causes the free angle to be used up between the spring **29** and the hub **11** in the reverse direction and the friction jump **R1** is set analogous with the other positive turning direction wherein an angular stagger is observed in the case of the two turning directions which results from the non-uniformity of the active areas in the pull and push type operation of the preliminary damper **2** (FIG. **6a**). With a reduction in the turning angle the first preliminary damper stage runs through the neutral phase and a negative turning moment M is built up up to the end **B** of the push direction.

FIG. **7** shows a detail of an embodiment with a torsion vibration damper **201** in which the input parts **205**, **207** are tensioned relative to one another by means of the diaphragm spring **233**, with axial interposition of the cone **231** which itself is axially supported on a radially extending shoulder **211a** of the hub **211** so that a centering of the disc part **207** on the cone **231** is effected by the axial spring constant of the diaphragm spring **233**. For optimisation of the centering of the side disc **207** on the cone **231** the cone angle α of the contact surface of the cone **231** is set between $0 < \alpha < 45^\circ$, preferably $25^\circ < \alpha < 35^\circ$. On a relative rotation between the hub **211** and the disc parts **205**, **207** a friction torque arises at the cone **231** which is determined in dependence on the cone angle α , the friction surfaces in contact with one another, the spring constant of the diaphragm spring **233** and the friction values of the relatively rotating parts. Thus the friction engagement between the region **207a** and the cone **231** at the cone surface **231a** and/or preferably between the cone **231** and the output of the energy store **219** of the pre-damper at the contact surface **231b** can be adjusted so that here between the two parts **231**, **219** a friction plate can be provided. The driven side control or biasing of the energy store **227** occurs by means of a control plate **227a** engaging from the side of the disc part **205** in the energy store **227**, which control plate engages in the tothing **219a** of the hub **211**.

FIG. **8** shows a further construction, similar to the embodiment of FIG. **7**, of a detail relating to the cone **331** with a cone angle $0 < \alpha < 45^\circ$, preferably $25^\circ < \alpha < 35^\circ$ and a friction contact to the hub tothing **219a** under formation of friction surface **331b**, which produces a friction torque on a relative rotation of the radially outer disc part **305**, **307** which are axially connected to one another, against the hub

311. The two disc parts **305**, **307** are thus tensioned, against the hub **311**, with axial interposition of the cone **331** on the one side and an abutment ring **332** on the other side by means of the axially operating diaphragm spring **333** which supports itself on the disc part **305** and on the abutment ring **332**.

FIG. **9** shows a modified embodiment of the torsion vibration damper **1** in FIG. **1**. The torsion vibration damper **401** of FIG. **9** illustrated as a part section has in the region of a pre-damper **402** a friction device **428** arranged so that the diaphragm spring **433** itself offers no friction function but only the tension of the friction control disc **429** against the cone **431** on the one hand as well as against the flange **408** on the other hand. In this way, a two stage construction of the friction device **438** (**428?**) is possible.

The first step is to find by an axial tensioning of the two radially outer disc parts **406**, **407** which are connected with one another, with axial inter position of the cone **431** with the hub **411** by means of the diaphragm spring **488**. By the pretensioning of the disc parts **405**, **407** there arises a friction engagement on a relative rotation of the hub **411** against the disc parts **405**, **407** as a first friction stage at the contact surface **431a** between the cone **431** and the disc part **407**, wherein on appropriate setting up of the friction conditions, the friction engagement in accordance with FIGS. **7** and **8** also in the contact region **431b** between hub **411** and cone **431** can be transferred, in that for example the cone angle α of the contact surface **431c** can be steeper wherein at this position the friction torque can be reduced and the centering of the disc part **405**, **407** on the cone **431** is improved.

The second friction stage occurs on a relative rotation of the flange **408** relative to the friction control disc **429**, thus in the working region of the pre-damper **402**, wherein the friction torque is formed at the contact surface **429a** of the friction control disc **429** to the flange **408** and the friction control disc **429** is hooked into the output part **419** and by means of a rotational play between the parts **429**, **419** a slipping friction can be produced. In order to avoid the diaphragm spring **433** from moving relative to the cone **431** and/or the friction control disc **429**, radially projecting extensions **433a**, **433b** are provided at the inner and outer circumferences, which with axially raised cams **431d** of the cone **431** and recesses **429b** of the friction control disc **429** form rotationally fast connections. The diaphragm spring **433** produces in the embodiment an increased tension of the cone **431** with the disc part **407** additionally to the action of the diaphragm spring **408**, whereby especially on a misalignment of the drive unit in the gear box an improved tensioning and thus a better centering of the disc part **405** on the cone and a better defined friction engagement is possible.

The patent claims filed with the application are proposed wordings without prejudice for obtaining wider patent protection. The applicant retains the right to claim further features disclosed up until now only in the description and/or drawings.

References used in the sub-claims refer to further designs of the subject of the main claim through the features of each relevant sub-claim; they are not to be regarded as dispensing with obtaining an independent subject protection for the features of the sub-claims referred to.

The subjects of these sub-claims however also form independent inventions which have a design independent of the subjects of the preceding claims.

The invention is also not restricted to the embodiments of the description. Rather numerous amendments and modifications are possible within the scope of the invention, particularly those variations, elements and combinations

and/or materials which are inventive for example through combination or modification of individual features or elements or process steps contained in the drawings and described in connection with the general description and embodiments and claims and which through combinable features lead to a new subject or to new process steps or sequence of process steps insofar as these refer to manufacturing, test and work processes.

What is claimed is:

1. Torsion vibration damper, more particularly for motor vehicle clutch discs, having at least one preliminary damper acting in a predetermined angular region and having energy accumulators of lower stiffness, and at least one main damper acting in a predetermined angular region and having energy accumulators of greater stiffness, wherein the energy accumulators are active between the respective input and output parts of the preliminary and main dampers, and the output part of the torsion vibration damper is a hub provided with inner profiled sections to fit onto a gearbox shaft, as well as with a flange with inner profiled sections forming the output part of the main damper, wherein the inner profiled sections engage with the outer profiled sections of the hub and through this profiling the flange of the main damper is able to execute restricted relative rotation relative to the hub, as well as having at least one disc part which forms the input part of the main damper and holds friction linings, at least one friction device, and one spring which engages in the outer profiled sections and controls at least one part of the friction device and defines the friction engagement.

2. Torsion vibration damper according to claim 1 wherein the hub is in two parts.

3. Torsion vibration damper according to claim 1 wherein a restricted relative rotation forming a free angle is possible between the spring and hub.

4. Torsion vibration damper at least according to claim 1, wherein the relative rotation between the spring and the hub takes place in a part of the angular area of the operating range of the energy accumulator of the preliminary damper.

5. Torsion vibration damper at least according to claim 1, wherein the relative rotation between the spring and the hub causes a delay in the friction engagement defined by the spring about the free angle α .

6. Torsion vibration damper at least according to claim 1 wherein the free angle α lies in the region of between $\pm 2^\circ$ and $\pm 3^\circ$, and preferably amounts to $\pm 2.5^\circ$.

7. Torsion vibration damper at least according to claim 1 wherein the spring has an inner profiled section complementary with the external profiled section of the hub.

8. Torsion vibration damper at least according to claim 1 wherein the external profiled section of the hub and the internal profiled section of the spring form teething which allows the free angle α .

9. Torsion vibration damper at least according to claim 1 wherein the output part of the preliminary damper is connected rotationally secured to the hub.

10. Torsion vibration damper at least according to claim 1 wherein the input part of the preliminary damper is formed as a friction device.

11. Torsion vibration damper at least according to claim 1 wherein the spring is tensioned between the input part of the preliminary damper and disc part and/or a component part fixedly connected to same.

12. Torsion vibration damper at least according to claim 1 wherein the component part connected fixedly to the disc part is a second disc part spaced apart through spacer bolts.

13. Torsion vibration damper at least according to claim 1 wherein the component part is a friction ring fixed on the second disc part.

14. Torsion vibration damper at least according to claim 1 wherein the spring has an external profiled section with at least one radially outwardly pointing tongue.

15. Torsion vibration damper at least according to claim 1 wherein the tongue(s) has or have radially outwards an approximately semi-circular shaped recess.

16. Torsion vibration damper at least according to claim 1 wherein the tongue or tongues spreads or spread out towards the radial outer side.

17. Torsion vibration damper at least according to claim 1 wherein the input part of the preliminary damper has a rounded end on the axial side facing the spring in the area of the contact face between the input part and the spring which is tensioned with a contact bearing angle.

18. Torsion vibration damper at least according to claim 1 wherein the rounded end has such a pitch angle that the bearing angle β of the spring is approximately $\beta=0$.

19. Torsion vibration damper at least according to claim 1 wherein the input part of the preliminary damper has at least one axially extending pin on the axial side facing the spring.

20. Torsion vibration damper at least according to claim 1 wherein the number of pins corresponds to the number of recesses on the tongues on the outer circumference of the spring.

21. Torsion vibration damper at least according to claim 1 wherein the pin (pins) engages (engage) with play in the recess of the tongue(tongues).

22. Torsion vibration damper at least according to claim 1 wherein the friction ring is fitted by at least one axially extending pin into a bore provided in the disc part.

23. Torsion vibration damper at least according to claim 1 wherein the friction ring has a ring located on the outer circumference, raised axially in the direction of the spring and forming an axial ring face.

24. Torsion vibration damper at least according to claim 1 wherein the resulting ring face drops towards its inner diameter.

25. Torsion vibration damper at least according to claim 1 wherein through the dropping ring face a phase angle γ is formed so that the contact bearing angle β of the spring on the friction ring is approximately $\beta=0$.

26. Torsion vibration damper at least according to claim 1 wherein the friction ring forms at least a part of the friction device of the preliminary and main damper.

27. Torsion vibration damper at least according to claim 1 wherein a further plate spring belonging to a friction device of the main damper is mounted outside the outer circumference of the raised ring of the friction ring.

28. Torsion vibration damper at least according to claim 1 wherein the plate spring is supported on axially aligned tabs of a friction control part which controls a part of the friction device of the main damper.

29. Torsion vibration damper at least according to claim 1 wherein the friction control part controls a second stage of the friction device of the main damper.

30. Torsion vibration damper at least according to claim 1 wherein the plate spring is supported axially against the radially inner non-raised ring face of the friction ring.

31. Torsion vibration damper at least according to claim 1 wherein the input part of the preliminary damper is fitted in windows provided for holding the energy accumulator in the input part of the main damper.

32. Torsion vibration damper at least according to claim 1 wherein correct fitting is produced through axially moulded pins provided for keyed engagement at the two radially inner corners of the window in the input part of the preliminary damper.

33. Torsion vibration damper at least according to claim 1 wherein the preliminary damper is housed axially between the disc parts.

34. Torsion vibration damper at least according to claim 1 wherein the external profiled section of the hub is continued in a second hub part and the spring engages by its inner profiled section into an external profiled section of the cone.

35. Torsion vibration damper at least according to claim 1 wherein the external profiled section of the hub differs from the external profiled section of the cone.

36. Torsion vibration damper, in particular according to one of the preceding claims, characterised in that the two disc parts are tensioned against the hub with axial interposition of the cone by means of an axially operating energy accumulator.

37. Torsion vibration damper at least according to claim 1 wherein a conical surface of the cone with a cone angle α forms a contact surface to one or both disc parts.

38. Torsion vibration damper at least according to claim 1 wherein the disc parts are centered on the cone.

39. Torsion vibration damper at least according to claim 1 wherein the cone angle α lies in the region $0^\circ < \alpha < 45^\circ$, preferably $25^\circ < \alpha < 35^\circ$.

40. A torsion vibration damper comprising

at least one pre-damper, operative over a first angular range, with a first input part and a first output part having an initial relative position to each other, and first energy accumulators being compressible between the first input part and the first output part,

at least one main damper, operative over a second angular range, with a second input part and a second output part, and second energy accumulators being compressible between the second input part and the second output part, the second energy accumulators having a higher degree of stiffness than the first energy accumulators,

a damper output part and a friction control disc,

a first diaphragm spring with an inner radius and an outer radius, wherein the friction control disc is tensioned by the first diaphragm spring against the flange, wherein the damper output part is constituted by a hub with an external profile, the second output part is constituted by a flange being received by the hub and having an internal profile that engages the external profile with angular play,

the second input part is constituted by two side discs that are constrained to have no rotational mobility relative to each other,

the first input part and the first output part are arranged axially between one of the two side discs and the flange, the first output part being rotationally constrained to the hub,

the friction control disc is arranged axially between the flange and another of the two side discs, the friction control disc engaging the first output part with play, whereby the main damper is activated at a point within the first angular range, but is made operative only subsequent to a relative angular displacement between the first input part and the first output part from their initial position relative to each other, and

wherein the first diaphragm spring and the friction control disc are constrained from rotating in relation to each other.

41. The torsion vibration damper of claim 40, wherein the friction control disc is in direct frictional engagement with the flange.

42. The torsion vibration damper of claim 40, further comprising a first friction ring, wherein the first diaphragm spring at the inner radius bears against the first friction ring.

43. The torsion vibration damper of claim 42, wherein the first friction ring is adjacent to and bears directly against one of the side discs.

44. The torsion vibration damper of claim 42, wherein the first diaphragm spring is axially tensioned by direct compressive contact at the outer radius with the friction control disc and at the inner radius with the first friction ring.

45. The torsion vibration damper of claim 42, further comprising at least one conical surface disposed between the hub and one of the side discs and serving to center said one of the side discs on the hub, wherein said conical surface is constituted by the first friction ring.

46. The torsion vibration damper of claim 40, further comprising at least one conical surface disposed between the hub and one of the side discs and serving to center said one of the side discs on the hub.

47. The torsion vibration damper of claim 40, wherein the pre-damper has a first characteristic curve divided into a first pre-damper stage and a second pre-damper stage, and the main damper has a second characteristic curve divided into a first main damper stage and a second main damper stage.

48. The torsion vibration damper of claim 40, wherein the pre-damper has a first friction stage comprising a second diaphragm spring bearing axially against one of the two side discs and bearing against the hub in a direction towards another of the two side discs.

49. The torsion vibration damper of claim 48, further comprising a second friction ring interposed between the second diaphragm spring and the hub.

50. The torsion vibration damper of claim 48, wherein the friction control disc and the second diaphragm spring are arranged on opposite sides of the flange.

51. A torsion vibration damper comprising

at least one pre-damper, operative over a first angular range, with a first input part and a first output part having an initial relative position to each other, and first energy accumulators being compressible between the first input part and the first output part,

at least one main damper, operative over a second angular range, with a second input part and a second output part, and second energy accumulators being compressible between the second input part and the second output part, the second energy accumulators having a higher degree of stiffness than the first energy accumulators,

a damper output part and a friction control disc,

a first diaphragm spring with an inner radius and an outer radius, wherein the friction control disc is tensioned by the first diaphragm spring against the flange, wherein the damper output part is constituted by a hub with an external profile, the second output part is constituted by a flange being received by the hub and having an internal profile that engages the external profile with angular play,

the second input part is constituted by two side discs that are constrained to have no rotational mobility relative to each other,

the first input part and the first output part are arranged axially between one of the two side discs and the flange, the first output part being rotationally constrained to the hub,

the friction control disc is arranged axially between the flange and another of the two side discs, the friction

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control disc engaging the first output part with play, whereby the main damper is activated at a point within the first angular range, but is made operative only subsequent to a relative angular displacement between the first input part and the first output part from their initial position relative to each other, 5

the first diaphragm spring and the friction control disc are constrained from rotating in relation to each other, and wherein the first diaphragm spring has projections at the outer radius and the friction control disc has recesses receiving the projections, whereby the first diaphragm spring and the friction control disc are constrained from rotating in relation to each other. 10

52. A torsion vibration damper comprising 15

at least one pre-damper, operative over a first angular range, with a first input part and a first output part having an initial relative position to each other, and first energy accumulators being compressible between the first input part and the first output part, 20

at least one main damper, operative over a second angular range, with a second input part and a second output part, and second energy accumulators being compressible between the second input part and the second output part, the second energy accumulators having a higher degree of stiffness than the first energy accumulators, 25

a damper output part and a friction control disc, 30

a first diaphragm spring with an inner radius and an outer radius, wherein the friction control disc is tensioned by the first diaphragm spring against the flange, 30

a first friction ring, wherein the first diaphragm spring at the inner radius bears against the first friction ring, wherein 35

the damper output part is constituted by a hub with an external profile, the second output part is constituted by a flange being received by the hub and having an internal profile that engages the external profile with angular play, 35

the second input part is constituted by two side discs that are constrained to have no rotational mobility relative to each other, 40

the first input part and the first output part are arranged axially between one of the two side discs and the flange, the first output part being rotationally constrained to the hub, 45

the friction control disc is arranged axially between the flange and another of the two side discs, the friction control disc engaging the first output part with play, whereby the main damper is activated at a point within the first angular range, but is made operative only subsequent to a relative angular displacement between the first input part and the first output part from their initial position relative to each other, and 50

wherein the first diaphragm spring is axially tensioned by direct compressive contact at the outer radius with the friction control disc and at the inner radius with the first friction ring. 55

53. A torsion vibration damper comprising 60

at least one pre-damper, operative over a first angular range, with a first input part and a first output part having an initial relative position to each other, and first energy accumulators being compressible between the first input part and the first output part, 65

at least one main damper, operative over a second angular range, with a second input part and a second output

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part, and second energy accumulators being compressible between the second input part and the second output part, the second energy accumulators having a higher degree of stiffness than the first energy accumulators,

a damper output part and a friction control disc, 5

a first diaphragm spring with an inner radius and an outer radius, wherein the friction control disc is tensioned by the first diaphragm spring against the flange, 10

a first friction ring, wherein the first diaphragm spring at the inner radius bears against the first friction ring, 10

at least one conical surface disposed between the hub and one of the side discs and serving to center said one of the side discs on the hub, wherein said conical surface is constituted by the first friction ring, wherein 15

the damper output part is constituted by a hub with an external profile, the second output part is constituted by a flange being received by the hub and having an internal profile that engages the external profile with angular play, 20

the second input part is constituted by two side discs that are constrained to have no rotational mobility relative to each other, 25

the first input part and the first output part are arranged axially between one of the two side discs and the flange, the first output part being rotationally constrained to the hub, 30

the friction control disc is arranged axially between the flange and another of the two side discs, the friction control disc engaging the first output part with play, whereby the main damper is activated at a point within the first angular range, but is made operative only subsequent to a relative angular displacement between the first input part and the first output part from their initial position relative to each other. 35

54. A torsion vibration damper comprising 40

at least one pre-damper, operative over a first angular range, with a first input part and a first output part having an initial relative position to each other, and first energy accumulators being compressible between the first input part and the first output part, 45

at least one main damper, operative over a second angular range, with a second input part and a second output part, and second energy accumulators being compressible between the second input part and the second output part, the second energy accumulators having a higher degree of stiffness than the first energy accumulators, 50

a damper output part and a friction control disc, wherein 55

the damper output part is constituted by a hub with an external profile, the second output part is constituted by a flange being received by the hub and having an internal profile that engages the external profile with angular play, 60

the second input part is constituted by two side discs that are constrained to have no rotational mobility relative to each other, 65

the first input part and the first output part are arranged axially between one of the two side discs and the flange, the first output part being rotationally constrained to the hub, 65

the friction control disc is arranged axially between the flange and another of the two side discs, the friction control disc engaging the first output part with play, 65

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whereby the main damper is activated at a point within the first angular range, but is made operative only subsequent to a relative angular displacement between the first input part and the first output part from their initial position relative to each other,

the pre-damper has a first friction stage comprising a second diaphragm spring bearing axially against one of

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the two side discs and bearing against the hub in a direction towards another of the two side discs, and wherein the friction control disc and the second diaphragm spring are arranged on opposite sides of the flange.

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